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14. ABSTRACT
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Forced Air Convection Thermal Switch Concept for Responsive Space Missions

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ABSTRACT: There has been a growing need in the Department of Defense to make space more responsive and cost effective. Instead of taking years to design and deploy a new satellite, the goal is weeks or even days. To meet this challenge, the methodologies used to design, manufacture, test, launch, and deploy satellites must radically change. One of the most challenging aspects of this problem is the satellite's Thermal Control System (TCS). Traditionally, the TCS is vigorously designed, analyzed, and optimized for every satellite mission. The ideal TCS for responsive space would be robust and modular with an inherent plug-and-play capability. The focus of this work was to investigate the design of a thermal control system based on a forced air convection thermal switch (FACTS) concept. The concept consists of separating the individual satellite subsystems and enclosing them each in hermetically sealed enclosures. The temperature is then controlled by modulating the heat transfer coefficient with a DC axial fan. Using FACTS, a conservative switching ratio of 69:1 was achieved.

INTRODUCTION

The 2001 Space Commission Report stated that "the United States (U.S.) is more dependent on space than any other nation"[1]. This is especially true for military applications where space is used for surveillance, communication, navigation, meteorology, theatre support, and force application. The U.S.'s use of existing space capabilities provides its forces an asymmetric edge during battle. Historically, large space assets have been considered strategic in nature because they take years to design, assemble, test, and deploy. A typical large satellite takes between three and ten years to design and field. In addition, the total mission cost ranges from hundreds of millions to billions of dollars. Compounding the problems are the significant cost and schedule overruns experienced by most programs. By their inherent nature, large complex systems are expensive and time intensive.

There has been a growing move in the aerospace industry and a growing need in the Department of Defense (DOD) to make space more responsive and cost effective. Instead of taking years to design and deploy a new satellite, the goal is weeks or even days. The DOD is actively pursuing the capability to make space operationally responsive. The goal is to extend the advantages space affords from the strategic planner to the battlefield commanders. The ability to launch a new space asset within days or hours of a battlefield commander's request will maintain the asymmetric advantage in future conflicts. Space provides the ultimate high ground, and Operationally Responsive Space (ORS) brings this advantage directly to the battlefield commander.

To meet this challenge, the methodologies used to design, manufacture, test, launch, and deploy satellites must radically change. For space to become operationally responsive, satellites must be easily manufactured, assembled, tested, and prepared for launch in a military depot style environment. Designs will have to be simple and robust so that Airmen play a central role and rather than Ph.D.-level scientists. Large geosynchronous satellites will continue to play an important role in space activities, but to achieve the goals of responsive space, components and systems will have to be standardized and simple, which translates to an increasing usage of small satellites.

One of the subsystems that will be challenging for the development of robust and modular architectures is the Thermal Control Subsystem (TCS). To design the TCS, virtually every aspect of the mission, the satellite, and the components must be known. The overall goal of the engineer is to reduce the mass of the system by trading cost and engineering time. As a result, every design is unique and requires extensive design, modeling, analysis, and test programs. For responsive space, the ideal TCS would be modular and robust to

accommodate the wide range of orbits, components, and payloads with minimal survival heater power. In addition, the design and assembly time must be dramatically reduced. The ultimate goal would be a plug-and-play TCS. Unfortunately missions, payloads, and requirements for ORS are still somewhat nebulous. As a result, bus architectures and specific components have not been identified, which make it difficult to derive even initial thermal system requirements.

One technology that is appealing for ORS missions is a thermal switch. Thermal switches provide thermal control by switching between high and low heat transfer regimes at a specific set point. When the temperature is below the set point, the switch is off, and its heat transfer is low. When the temperature is above the set point, the switch turns on and closes the heat transfer path. For passive conduction based thermal switches, this is typically done by placing the hot side and cold side of the switch in intimate contact. When the temperature of the component drops below the set point, the surfaces are separated, and conduction is minimized.

Thermal switches are an optimal solution for ORS because of the flexibility they provide. When mounted between the structure and the component, thermal control of each component can be decoupled. Different set points can be used for different components and applied only to components that need it. In addition, because the thermal switch minimizes heat transfer in the off position, radiators can be oversized, multi-layer insulation (MLI) can be eliminated, and survival heater power can be significantly reduced. Using heat switches results in a completely different design approach than traditional methodologies.

Although they have important advantages, thermal switches have never been used for whole satellite thermal control, but rather for niche cryogenic sensor applications. One reason is they are limited significantly in size and capability as an inherent property of how they function. To work, they must minimize conduction when "off", which means an absolute minimal mechanical support. When switched "on", they need maximum contact conduction. These two opposing requirements have caused failures that have prohibited their use in general. Another reason heat switches have not seen wide spread use is that they add a thermal resistance to the heat path. Adding a heat switch adds another interface to the design. Ultimately, this impedes the effectiveness of the radiators by increasing the temperature rise from the radiator to the component. For systems that are already operating near the limits of the radiator, the additional interface will cause the component to exceed their upper temperature limit requiring the radiator surface area to be increased. There are still significant design challenges for thermal switches. However, instead of using conduction-based heat switches, a forced convection based heat switch concept was investigated.

DESCRIPTION OF THE FACTS CONCEPT

The focus of this work was to investigate the design of a thermal control system based on a Forced Air Convection Thermal Switch (FACTS) concept. The concept consists of separating the individual satellite subsystems and enclosing them each in hermetically sealed enclosures as shown below. A fan is then used to control the heat transfer rate from the components to the base plate of the enclosure and ultimately to the radiator panels. By separating the subsystems, the capabilities of the bus can be modified and tailored by swapping out different subsystem enclosures. It also simplifies the overall design of the TCS because it limits the number of interfaces that must be controlled. Using sealed enclosures and forced air convection is not a completely new concept. It has been used before to cool electronics at high altitudes where there is not enough air for adequate cooling[2]. In addition, some Russian satellites have used sealed satellites and air convection as their primary TCS.

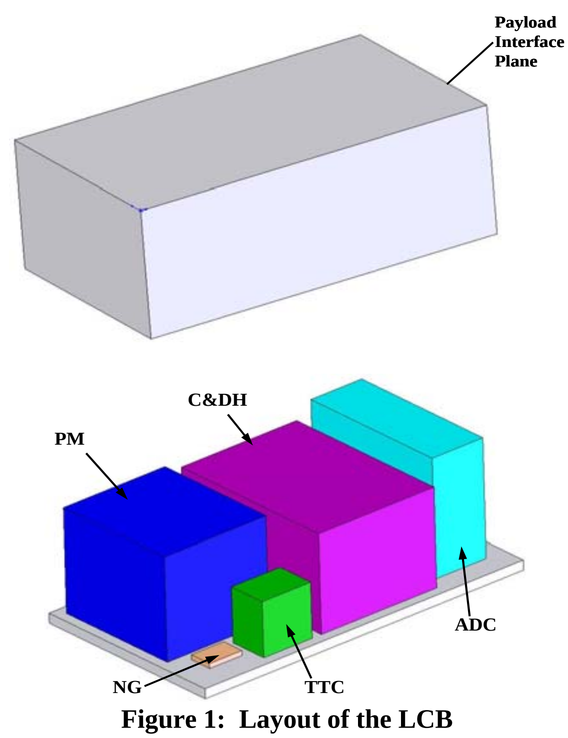

Figure 1: Layout of the LCB

There are significant advantages and disadvantages to this type of system. Forced convection provides higher

heat transfer coefficients than conduction. Therefore, a more efficient design is possible, and the thermal gradient over the subsystem can be reduced. Reducing the thermal gradient across the components reduces thermal stress. Secondly, a simple DC axial fan can be used as a thermal switch. When heat loads are high, the fan is switched on and provides additional cooling through convection. When loads are reduced, the fan is turned off, and heat is only transferred through conduction and radiation. The result is a reduction in survival heater power. Finally, sealing the enclosure provides significant advantages for depot-style operations. It reduces the cleanliness requirements for warehouse storage. Also, the importance of contamination and thermal joint quality requirements are reduced since the components are sealed in an atmospheric environment. Of course, there are significant challenges that must be addressed.

The biggest disadvantage to a forced convection system is the added mass required to maintain an internal pressure of 1 atm in the hard vacuum of space. Sealing the box to prevent leakage and eventually failure is also a critical design factor. Finally, adding a fan increases both the power requirements of the bus and the complexity of the ADC subsystem. A standard DC axial fan capable of producing a flow rate of 30 CFM against a pressure of 6 mmH_2O requires approximately 4 W of power. This adds stress to the power system and an added load to the TCS. The complexity added to the ADC subsystem is the addition of a rotating component with its own vibration spectrum that turns off and on almost instantaneously. However, the advantage of a modular, robust system outweighs the disadvantages when a short turn-around-time becomes more important than mass.

THERMAL REQUIREMENTS

Before the design, modeling, and analysis of a TCS can commence, a certain level of fidelity of the bus design is needed before the basic requirements for the thermal control subsystem can be identified. Unfortunately, requirements for ORS missions are somewhat nebulous; however, there is one assumption that can be made. Because of launch vehicle limitations, ORS missions will likely be relegated to 450 kg class satellites. Using this basic assumption, the capabilities that a small satellite bus can provide can be determined. In a previous effort, two satellite busses were sized to meet responsive space needs[3]. The first bus provided minimal capabilities, while the other provided significantly improved capabilities. These two busses represent a lower and upper bounds for design and are summarized below.

Table 1: Satellite System Summary LCB

Subsystem	Capability	Mass	Power	Size
		[kg]	[W]	[cm]
Attitude Determination & Control (ADC)	1°-5° attitude control	10.3	18.5	30 x 24 x 12
Telemetry, Tracking, & Command (TTC)	1 Mbs, S-band transmitter	2.8	7.4	9.8 x 9.6 x 7.2
Navigation & Guidance (NG)	12 channel GPS receiver	0.02	0.8	7.0 x 4.5 x 1.0
Command & Data Handling (CDH)	Plug-n-play USB architecture	15.2	50	34 x 25 x 20
Power Management (PM)	500 W, 3J array, PPT system	18.3	70.3	25 x 23 x 21
Structure	Al Honeycomb Panels	21.5	n/a	27 x 40.5 x 71
Propulsion	No propulsion system	0	0	0 x 0 x 0
		68.1	**147.0**	**27 x 40.5 x 71**

Table 2: Satellite System Summary HCB

Subsystem	Capability	Mass	Power	Size
		[kg]	[W]	[cm]
Attitude Determination & Control (ADC)	0.1°-1° attitude control	23.3	49.5	35 x 35 x 22
Telemetry, Tracking, & Command (TTC)	274 Mbs, Ku-band transmitter	10.6	64.4	25 x 25 x 15
Navigation & Guidance (NG)	12 channel GPS receiver	0.0	0.8	7.0 x 4.5 x 1.0
Command & Data Handling (CDH)	Plug-n-play USB architecture	15.2	50	34 x 25 x 20
Power Management (PM)	1500 W, 3J array, PPT system	54.6	253	72 x 23 x 21
Structure	Al Honeycomb Panels	38.6	n/a	52 x 40.5 x 71
Propulsion	Not applicable	0	0	0 x 0 x 0
		142.32	**417.7**	**52 x 40.5 x 71**

For the focus of this paper, the capabilities of the bus are somewhat inconsequential. The key parameters here are the power and size for each subsystem and the overall bus. With this information, first level requirements for the TCS can be defined, and overall system architectures can be evaluated. For example, the overall size of the bus yields the radiator space available for heat rejection and, when coupled with the system's surface properties, provides the external heat load that must be managed. Also, the size and power for each of the subsystems provides the base plate area available to transfer heat from the components to the satellite bus and the power density that must be managed at a subsystem level.

In addition to the above, the location and orientation of the components must be determined. To simplify the integration of the components into the bus, they were separated by subsystem and sealed in enclosures. As noted before, this provides two advantages. The first is storage in a depot-style environment. The second is simplifying interface standards. Because of this separation, the thermal design can be separated into two parts: overall bus design and component specific design. At the interface between the bus and the subsystems, a natural breakpoint occurs. Rather than having to specify interface standards for every type of component, standards would only have to be created for the subsystem enclosure/bus interface. By separating at that location, the subsystem supplier would be responsible for developing the thermal design of the components inside the enclosure; whereas, the system integrator would be responsible for developing the overall thermal control of the bus. The interface between the bus and the subsystems enclosures would be dictated by a thermal design standard that both parties would have to follow. For this analysis, the interface conductivity between the enclosure and the bus was specified as 435 W/m^2K, which represents a perimeter bolt pattern and an RTV interstitial material[4].

Figure 1, above, shows the location and orientation of the subsystem enclosures for the bus. In addition, the figure shows the face that is reserved as the interface plane between the bus and the payload. At this location there is no heat transfer between the bus and the payload or between the bus and the external environment.

THERMAL MODELING APPROACH

Using the characteristics for each subsystem, a thermal model was developed for the two buses. The overall bus structure was model with 1" thick aluminum honeycomb panels. All of the individual subsystems, with the exception of the Command and Data Handling (CDH) subsystem, were modeled as aluminum enclosures with a uniformly distributed heat load. This approach was taken to develop a model that provided enough fidelity for an accurate thermal balance but was not processing time intensive. Since the focus of this effort was to develop a robust TCS design based on the FACTS concept, the CDH subsystem was modeled in detail. It was used as the primary subsystem for analysis for the FACTS concept.

The CDH subsystem consisted of four Printed Circuit Boards (PCBs) mounted to a backplane PCB. The boards were modeled as 0.3 cm thick PCBs fabricated out of FR4 2 oz copper. An edge contact conductivity of 17.7W/m-K was used for the connection between the PCBs and the mounting rails. All of the boards were mounted on 0.5 cm thick aluminum rails to conduct heat to the walls of the enclosure. Finally, with the exception of the processor, the heat loads were applied as uniform loads over the board. The load on the back plane and legacy interface boards was 5 W. It was 10 W for the power management and the processor board. In addition, a processor heat load of 10 W was applied to a 2 cm by 2 cm area on the processor board. The total power consumption for the system was 50 W and the base plate area was 25" x 34". The location and orientation of the PCBs are shown on Figure 2. The figure also shows the flow path the was modeled for the system with the fan located between section 1 and 2.

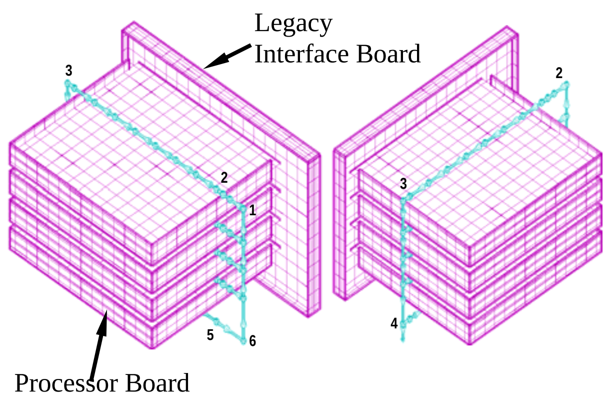

Figure 2: Schematic of the CDH System

OPTIMIZING THE ENCLOSURE DESIGN FOR THE FACTS CONCEPT

In the FACTS concept, the fan provides two distinct functions. The first function is to increase the heat transfer rate through the subsystem to keep the component temperatures below their maximum operating temperatures. Second, the fan functioned as a heat switch. For the hot case, the fan provides

additional cooling to increase the heat transfer rate of the subsystem. During the cold case, the fan is switched off, and heat is primarily transferred by conduction through the enclosure. The result is a significant reduction in survival heater power. The keys to TCS design using thermal switches is to maximize heat transfer during the hot case and isolate the system during the cold case so that it retains its heat. For the hot case, heat is transferred by convection, conduction through the enclosure, and radiation from the enclosure to the interior of the satellite. For the cold case, convection is eliminated, but the conduction and radiation paths are still present. The following sections look at optimizing the system for each of these cases.

Maximizing the Convective Heat Transfer to the Plate

Since it is important to maximize the convective heat transfer for the hot case, two designs cases were evaluated. For the first case which was the base line case, a bare aluminum surface was used to transfer heat from the fluid to the base plate. The advantage of using a bare aluminum base plate is its simplicity. The disadvantage is the relatively low heat transfer coefficient. For the second case, a finned heat exchanger (HX) was added to the aluminum base plate. Adding the finned HX significantly improves convective heat transfer, but it also increases the complexity, mass, and cost of the system. In the design of finned HXs, the goal is to increase the heat transfer coefficient to the point that adequate cooling is obtained. This is typically done by reducing the cross sectional area of the channels through the exchanger. The tradeoff is an increase in the pressure drop of the system. The final design consisted of two rows of 1 cm tall fins. The thickness of each fin was 1 mm, and the spacing between fins was 0.5 cm. A schematic of the heat exchanger is shown in Figure 3.

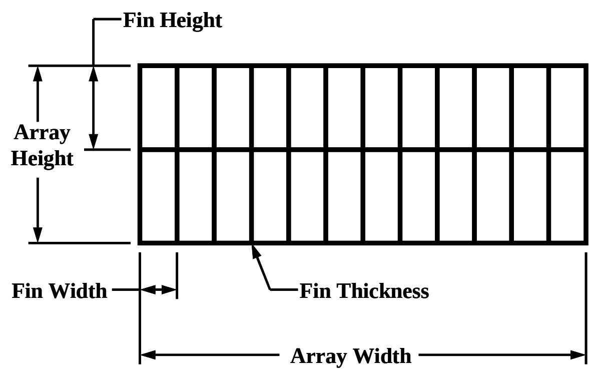

Figure 3: Schematic of the Finned HX Design

To determine the effect of the finned heat exchanger, the convection heat transfer coefficient was plotted as a function of flow rate and is shown in Figure 4. To determine the convection coefficient for the bare base plate design, a channel height of 0.03 m and a channel width of 0.25 m were used. In the baseline design, the air gap between the base plate and the components inside the enclosure was on the order of 3 cm, the flow was laminar for flow rates less than ~13 CFM and never reached fully turbulent conditions even when the flow rate was increased to 40 CFM. Heat transfer to the base plate would be significantly improved if fully developed turbulent flow conditions could be reached. The Reynolds number can be increased by either increasing the flow rate or by decreasing the cross sectional area of the flow channel. By inserting a finned heat exchanger into the flow path, the cross sectional area of the flow path is effectively reduced by forcing the fluid down smaller independent channels. Finned heat exchangers also improve heat transfer by increasing the surface area available for transfer. As a result, the convection coefficient was significantly higher for the finned HX design option, which increased by two orders of magnitude.

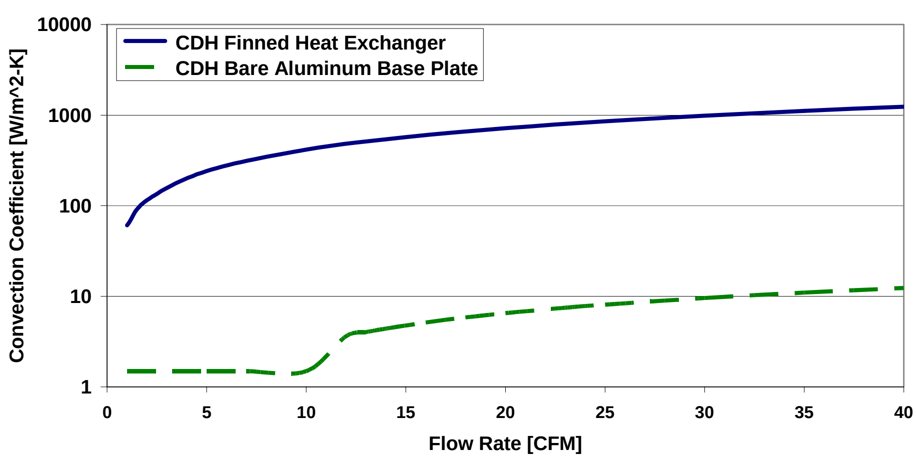

Figure 4: Effect of the Finned HX has on the Convection Coefficient

From the figure, the distinct flow regions for the bare aluminum base plate can be identified. For flow rates less than 10 CFM the flow is laminar making the Nusselt number and the convection coefficient constant. Above that point, the flow region transfers to the transition region and the Nusselt number increases with the Reynolds number. Even at 40 CFM the Reynolds number is only 8841, which means fully turbulent flow never develops. As for the finned heat exchanger, the flow is in the transition region even at flow rates as low as 1 CFM, and the flow is fully turbulent for flow rates above 3 CFM. By adding the finned heat exchanger, the heat transfer from the fluid to the base plate is significantly improved.

The effect on the overall subsystem was investigated next. For the CDH subsystem, the overall effect on the maximum temperature is shown on Figure 5. For the bare aluminum base plate design, the maximum temperature of the subsystem follows the same general behavior as the convection coefficient. For flow rates less than 10 CFM, the temperature is not quite constant but has a very small negative slope. The difference is the result of the different flow behaviors in the other parts of the subsystem. For example, the spacing between the PCBs is much smaller than between the components and the base plate. As a result, the flow characteristics will be slightly different. Above 10 CFM, the maximum temperature decreases more rapidly with increasing flow rate because of the change to the transition region.

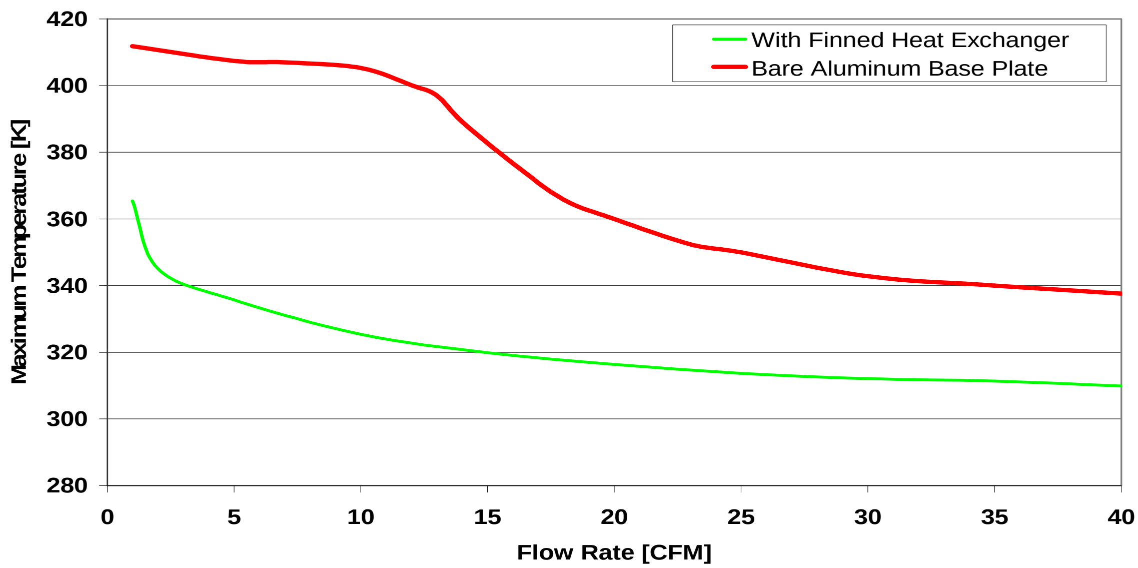

Figure 5: Effect of the Finned HX on the Maximum Temperature for the CDH System

As for the finned heat exchanger design, the behavior of the maximum temperature is consistent with the change in the convection coefficient. It is also important to note from the chart that there is a point of diminishing returns at approximately 11 CFM. At this point, the convection coefficient becomes greater than 435 W/m^2-K, which is the interface conductivity between the base plate and the electronics shelf. The additional reduction in the maximum temperature as the flow rate is increased up to 40 CFM is the result of the fan reducing the hot spot temperature and isothermalizing the subsystem.

In addition to increasing the flow rate, the isothermalizing effect can be improved by adding arrays of fins to the other walls of the enclosure. By adding fins to all of the walls of the enclosure, the heat transfer from the components to the fluid is increased because the overall heat transfer surface area is increased. For example in the CDH subsystem, the PCBs transfer heat to the fluid directly by convection. In addition, heat is transferred from the PCBs to the walls of the enclosure where it is also transferred to the fluid. By adding fins to the walls and increasing the heat transfer between the enclosure and the fluid, the overall heat transfer from the components to the fluid is increased. As a result, thermal design aspects of the components, such as the PCB spacing, become less critical. The effect of adding a 2 cm tall by 1 cm wide array of fins to all of the available enclosure walls is shown on Figure 6.

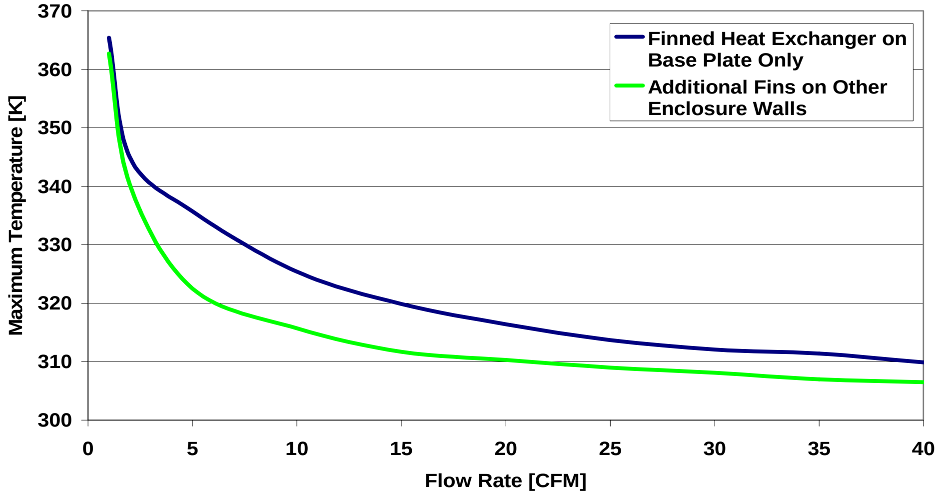

Figure 6: Effect Adding Arrays of Fins has on the Maximum Temperature

Adding fins to the other walls of the enclosure reduced the maximum temperature of the subsystem by an average of 6.1 K. The maximum reduction occurred at 5 CFM where the difference between the two was 13.2 K. Again, the system shows a point of diminishing returns, which occurs at approximately 15 CFM. Because of the enhanced heat transfer, the baseline enclosure design will include fins on the walls of the enclosure, and the baseline flow rate is 15 CFM. If additional cooling is needed, the flow rate could be increased to 40 CFM, but an easier solution would be to reduce the temperature at the subsystem mounting interface, i.e. increasing the size of the radiator.

Minimizing Heat Transfer for the Cold Case

For the hot case, the critical design parameter is maximizing convective heat transfer from the components to the base plate. As for the cold case, the critical design parameter is minimizing heat transfer through the system. This is accomplished by minimizing radiation from the enclosure to the bus and conduction from the enclosure to the base plate. Of the two, radiation exchange inside the bus is the easier one to negate. Using Multi-Layer Insulation (MLI), radiation between the enclosures and the bus can essentially be eliminated. However, MLI is expensive and difficult to work with. As an alternative, either bare aluminum enclosures or a low emissivity surface coating provides a more manageable solution. The emissivity of aluminum is 0.03. For components or scenarios were an optimal solution is required, MLI can be used, but for more situations bare aluminum provides acceptable performance.

Conduction from the enclosure to the base plate is much more difficult to minimize. The joint between the enclosure and the base plate is critical because it must hermetically seal the enclosure as well as isolate it from the base plate. Ideally, the joint would completely isolate the enclosure from the base plate making convection to the base plate and radiation between the enclosure and the interior surfaces of the satellite the only means of heat transfer. Since complete thermal isolation while maintaining a hermetic seal is not possible, the effect of the interface conductivity on the component temperature for the worst cold case was determined.

The temperature rise through the interface between the base plate and the enclosure is calculated with the equation below:

$$T_H - T_C = \Delta T = \frac{Q}{AK_J} \tag{1}$$

where T_H is the temperature on the hot side of the joint [K], T_C is the temperature on the cold side of the joint [K], Q is the heat load [W], A is the contact area [m^2], and K_J is the joint conductivity [W/m^2-K]. The joint conductivity for a bare interface is simply the contact conductivity. However, if an interstitial material is present, the joint conductivity is also affect by the thickness and the thermal conductivity of the interstitial material. Since the two interfaces and the interstitial material are in series, their thermal resistances are added. This is analogous to electrical resistances and the same rules apply. Figure 7 provides a schematic for clarity.

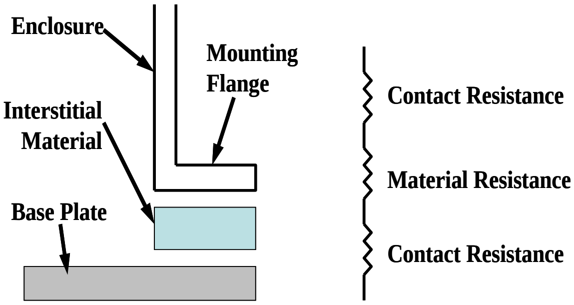

Figure 7: Schematic of the Thermal Joint between the Enclosure and the Base Plate

Using the electrical resistance analogue, the total resistance for the joint shown above is:

$$R_{Tot} = \frac{1}{A_{cont}}\left(\frac{2}{K_{\text{int}}} + \frac{L}{K}\right) \qquad (2)$$

where R_{Tot} is the thermal resistance [K/W], Acont is the contact area [m2], Kint is the interface conductivity [W/m-K], L is the base plate thickness [m], and K is the material conductivity [W/m-K]. The joint conductivity, K_J, is the inverse of the total joint resistance divided by the contact area.

Since the temperature on the hot side of the interface is dependent on the system parameters i.e. the contact area, the internal power dissipation, and the cold side temperature, it is difficult to identify a single joint conductivity that would meet the thermal needs for all potential components/subsystems. A very small joint conductivity on the order of 1 W/m^2-K would probably meet the needs of the majority of the components, but it might not be possible to design a thermal joint with that small of a thermal conductivity and still provide a hermetic seal. To better gauge the subsystem needs, the LCB and HCB designs were evaluated. Based on a simple energy balance, the cold case temperature for the LCB was 187.9 K. For the HCB, it was 183.0 K. Using the cold case power consumptions and the contact areas for each enclosure, based on the thermal joint above, the joint thermal conductivity required to keep the subsystem temperatures above the lower temperature limit of 273 K was calculated. The results are presented on Table 3.

To meet the needs of all of the subsystems on Table 3, a joint conductivity of 5 W/m^2-K is required; however, this does not take into account the temperature rise from the enclosure to the component. For subsystems with a higher power density, the component temperatures for the worst cold case could easily exceed the upper temperature limit if a thermal joint with a conductivity of 5 W/m^2-K is used. For those cases, the joint does not have to be replaced with a higher conductivity joint. Instead the fan, operating at a lower flow rate than the hot case scenario, would ensure the component temperatures do not exceed the upper temperature limit.

Table 3: Joint Conductivity Required to Meet the Minimum Temperature Limit

System	Heat Load [W]	Surface Area [m^2]	Power Density [W/m^2]	K_J [W/m^2-K]
LCB				
ADC	18.5	0.0168	1101.19	12.80
CDH	13.0	0.0236	550.85	6.41
PM	16.2	0.0184	880.43	10.24
TTC	7.4	0.0067	1101.19	12.80
HCB				
ADC	18.5	0.0228	811.40	9.43
CDH	13	0.0236	550.85	6.41
PM	41.2	0.0372	1107.53	12.88
TTC	7.4	0.016	462.50	5.38

Since eliminating the need for survival heater power for most cases depends on a low conductivity, hermetically sealed thermal joint of approximately 5 W/m^2-K, it is important to determine if such a joint is possible. Most hermetically sealed joints use a bolted joint with either an o-ring seal or a Teflon energized seal to provide a seal in a vacuum. To thermally isolate this type of joint, a low conductivity gasket is required. Using felt as an interstitial material, joint conductivities as low as 10 W/m^2-K are possible[4]. The joint conductivity can further be reduced by adding a 5 mm thick low conductivity Teflon spacer to the joint. The thermal conductivity of Teflon is 0.27 W/m-K. The resulting thermal joint conductivity is on the order of 4.8 W/m^{2}K. The bolts will also have to be isolated from the system by using low conductivity Teflon washers and sleeves.

In addition to the conductivity of the joint, there is another issue that must be addressed, which is the possibility that the air inside the enclosure will act as a thermal short when the fan is turned off. Since the satellite will be in a microgravity environment, natural convection can be ignored and only conduction through the gas must be considered. The conductivity of air at standard temperature and pressure (T = 25 C, P = 1atm) is 0.03 W/m-K. This is an order of magnitude smaller than the conductivity for any joint interstitial material. As long as an air gap of at least 2 cm is maintained between the components and the base plate, the heat transfer through the air can be ignored.

SUBSYSTEM DESIGN USING FACTS

The preceding analysis and discussion focused on the design of the subsystem enclosure to enhance the thermal switching effect of the fan. Specific attention was paid to the CDH subsystem to provide insight into the interactions of the design variables, but the goal was a more general development of the enclosure design. The importance of conduction through the interface, radiation between the enclosures and the interior of the satellite, and convection within the system were explored. The attention now turns to assessing the conductance ratio of the subsystem design

For the hot case, the amount of heat that can be rejected from the system is dependent on the heat transfer to the finned heat exchanger and the interface conductivity of the base plate. The heat transfer from the components to the air stream is also important, but it is component design dependent and will not be discussed. As for radiation exchange between the enclosure and the interior of the satellite the effect of radiation heat transfer is minimal for the hot case because of the low emissivity value for aluminum and will be ignored.

By using the convection heat transfer coefficient and the joint contact conductivity, the amount of heat that can be rejected from the system can be determined by noting that the heat transfer path is in series, and the thermal resistances are added. There are three resistances that must be considered. In addition to the two mentioned above, the thermal resistance through the base plate material must also be considered. The total resistance is determined with the following equation:

$$R_{Tot} = \left(\frac{1}{C_{FHE}Ah}\right)_{conv} + \left(\frac{L}{AK_{Al}}\right)_{base} + \left(\frac{1}{AK_J}\right)_{joint} \quad (3)$$

where C_{FHE} is a multiplier for the surface area added by the finned heat exchanger, A is the base plate area [m^2], h is the convection coefficient [W/m^2-K], L is the thickness of the base plate [m], K_{Al} is the conductivity of aluminum [W/m-K], and K_J is the joint conductivity [W/m^2-K]. C_{FHE} was determined with the equation below and is dependent on the heat exchanger design. For the design discussed above, the value of the multiplier was 5.8.

$$C_{FHE} = 2\left(\frac{h_{fin}}{S_{fin}} + \left(1 - \frac{t_{fin}}{S_{fin}}\right)\right) \quad (4)$$

Here, h_{fin} is the height of the fins [m], S_{fin} is the pitch of the fins [m], and t_{fin} is the thickness of the fins [m]. From Eq. 3 the conductance on a per area basis can be determined by transferring the base plate area to the left side of the equation and noting that the conductance is the inverse of the resistance. The equation in final form is below.

$$C_{Tot} = \frac{1}{R_{Tot}A} = \left[\left(\left(\frac{1}{C_{FHE}h}\right)_{conv} + \left(\frac{L}{K_{Al}}\right)_{baseplate} + \left(\frac{1}{K_J}\right)_{interface}\right)\right]^{-1} \quad (5)$$

The total conductance is dependent on the convection coefficient, which is dependent on the flow rate. For this system there are two flow rates of primary interest. These are the baseline flow rate (15 CFM) and the maximum flow rate (40 CFM). The input values are summarized below on Table 4. The conductance for the FACTS enclosure design for the hot case is 342 W/m^2-K at 15 CFM and 362 W/m^2-K at 40 CFM.

Table 4: FACTS Enclosure Design Input Values

Parameter	Variable	Value	Units
FHE Area Multiplier	C_{FHE}	5.8	
Convection Coefficient at 15 CFM	h	571.8	W/m^2-K
Convection Coefficient at 40 CFM	h	1236	W/m^2-K
Base Plate Thickness	L	0.006035	m
Conductivity of Aluminum	K_{al}	185	W/m-K
Base Plate Interface Conductivty	K_J	435	W/m^2-K

As for the cold case, conduction through the low conductivity joint is the most important heat transfer mechanism. For systems that are operating at the edge of the lower component temperature limit, a small amount of survival heater power should be added to the subsystem for safety. Otherwise, radiation exchange can be ignored, and only conduction through the low conductivity joint must be considered. Therefore, for the cold case, the total conductance for the FACTS enclosure design is 5 W/m^2-K.

Unfortunately, determining the conductance ratio is not as simple as dividing the hot case total conductance by the cold case total conductance because the surface area changes between the two. For the hot case, heat is transferred over the entire area of the base plate; whereas, it is only transferred through the mounting flange for the cold case. However, to provide a conservative estimate of the conductance ratio, it will be assumed that that the 5W/m^2-K total conductivity applies to the entire area of the base plate. Using this approximation, the conductance ratio is 69:1 for a flow rate of 15 CFM and 72:1 for a flow rate of 40 CFM. These values are comparable to paraffin based heat switches.

CONCLUSIONS

One of the most challenging aspects to Operationally Responsive Space is the satellite's Thermal Control Subsystem (TCS). Traditionally, the TCS is vigorously designed, analyzed, and optimized for every satellite mission. This "reinvention of the wheel" is costly and time intensive. The next generation satellite TCS must be robust, modular, and scalable in order to meet the needs of a wide range of missions, payloads, and thermal requirements. To address these issues, a thermal switch approach based on forced air convection was considered.

After an initial investigation into the feasibility and performance of a forced air convection thermal switch, it was shown that after the system was optimized for convection a conductance ratio on the order of 69:1 was achievable. The result is a more robust TCS and a significant reduction in survival heater power. In addition to the advantages provided by a thermal switch based design, the FACTS approach has other advantages compared to conduction-based heat switches. Problems associated with cold welding, material fatigue failure, and surface cleanliness are eliminated. Also, whereas conduction-based heat switches are inherently limited in size by their design, the FACTS approach is not and is best suited for subsystem implementation.

As with most systems, the advantages must be traded with the disadvantages. The disadvantages of the FACTS approach include the added system mass for the pressurized enclosure and the finned heat exchanger; the challenge of a reliable, thermally isolated, hermetically sealed enclosure; and the added complexity of an active thermal control system. As with most thermal switches, single point failures are inherent in the design. Fan failure or seal leakage would more than likely result in a catastrophic failure of the subsystem.

Taking the advantages and disadvantages into consideration, the FACTS approach is a viable solution for ORS and should be further investigated. The advantage of a modular, robust system outweighs the disadvantages when a short turn-around-time becomes more important than mass. Finally, it must be noted that the FACTS approach is only suitable for short duration mission.

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Nomenclature

A = base plate area [m^2]
A_{cont} = contact area [m^2]
C_{FHE} = surface area multiplier for the finned heat exchanger
h = convection coefficient [W/m^2-K]
h_{fin} = height of the fin [m]
K = material conductivity [W/m-K]
K_{Al} = conductivity of aluminum [W/m-K]
K_{int} = interface conductivity [W/m^2-K]
K_J = joint conductivity [W/m^2-K]
L = base plate thickness [m]
Q = heat generation [W]
R_{TOT} = total joint resistance [K/W]
T_C = cold side temperature [K]
T_H = hot side temperature [K]
ΔT = temperature change [K]
S_{fin} = pitch of the fins [m]
t_{fin} = fin thickness [m]